Spread in modal data obtained from wind turbine blade testing

Mladen Gibanica, Anders T. Johansson, Sadegh Rahrovani, Majid Khorsand and Thomas Abrahamsson
Chalmers University of Technology, Department of Applied Mechanics
SE-412 96 Göteborg, Sweden
e-mail: **gibanica@student.chalmers.se**

ABSTRACT

This paper presents a pre-study for an on-going research project in experimental dynamic substructuring, initiated by the SEM Substructures Focus Group. The focus group has selected a small wind turbine, the Ampair 600W, to serve as test bed for the studies. The turbine blades are considered in this study. A total of 12 blades have been tested for modal properties in a free-free configuration. The data has been acquired and analysed by students participating in the undergraduate course ”Structural Dynamics - Model Validation” at Chalmers University of Technology. Each blade was tested by different students as part of their required course work to account for spread in modal properties between the blades. A subset of the blades were tested independently multiple times to account for variability in the test setup. Furthermore, correlation analysis of test data was made with Finite Element model eigensolution data of the blade.

Keywords: substructuring, wind turbines, vibration testing, modal analysis, statistics.

1 Introduction

Advances in computer technology have made computer simulations of complicated physical objects possible in industry. More and more companies are relying on relatively cheap Finite Element (FE) modelling rather than expensive prototyping. However, a production sequence will always contain uncertainties, causing a spread between individuals in a produced population. This spread stems from a variety of sources, such as material spread, production process variability and finite precision in the assembly of components. To create models that account for these differences requires data about the spread, in terms of characteristic properties of the population. Modal analysis can be used in order to quantify the spread in dynamical properties, see Friswell and Mottershead [1]. Also, static measurements are of interest to quantify mass and geometrical properties.

In recent years, interest in experimental dynamic substructuring have lead SEM to create a focus group with the intention to increase the knowledge in experimental substructuring and theory. The group have chosen a common testbed, the Ampair 600W wind turbine, for benchmarking, see [2].

This paper deals with the blades of said wind turbine, which have previously been studied by, e.g. Harvie and Avitabile [3] and Nurbhai and Macknelly [4]. Vibration testing, mass and geometric measurements were performed on 12 blades. Emphasis is put on the spread between individual blades and between groups of blades. The lab testers were mainly students attending the course Structural Dynamics Model Validation at Chalmers University of Technology during the spring of 2012. During the course, each student validated and calibrated a rough FE model with their dynamical measurements. In this paper, one of the correlated models is presented.

2 Measurements

Vibration testing was performed on blades hung in a free-free like configuration. A total of 12 blades were tested, of which some were tested multiple times by different individuals. Multiple tests are structured in three rounds. All blades were tested

in round 1, while only some were tested in round 2 and 3. The static measurements consisted of characterising the geometry by measuring pitch angles at three cross-sections of the blade and weighing the blades.

2.1 Vibrations Test

Each student would prepare their assigned blade individually by first weighing the blade and then set it up for vibration testing. The whole procedure was supervised by teaching assistants. The blades were hung in thin and light-weight cords through two of the three bolt mounts in order to obtain an almost free-free configuration. Light-wight accelerometers of IEPE (Integrated Electronics Piezo Electric) type were used (PCB 352C22, 0.5 gramme) and were attached by synthetic wax at predefined locations, see Fig. 1. The predefined accelerometer locations were specified in [3]. Additional mass-loading caused by the measurement equipment was neglected. In total 21 accelerometers were used, 20 at the predefined locations, and 1 behind the shaker. The accelerometer behind the shaker was used to verify that the sensor at location 3 served as a direct accelerance. The shaker was placed just beside location 3, see Fig. 2b. The input signal was a stepped sine signal ranging from 30 Hz to 500 Hz with a step size of 0.25 Hz. In order to obtain modal properties, system identification of the obtained data was performed in MATLAB using the System Identification Toolbox. A linear model identification, N4SID, which is based on a state-space subspace method, further explained in [6], was used to estimate a model from the measured data. During the course each student made an individual system identification of their data, where parameters of the identification method differed. In preparing this paper, a new system identification was performed so that the same parameters were used for all blades. Thus, only the measurements are dependent on the lab testers.

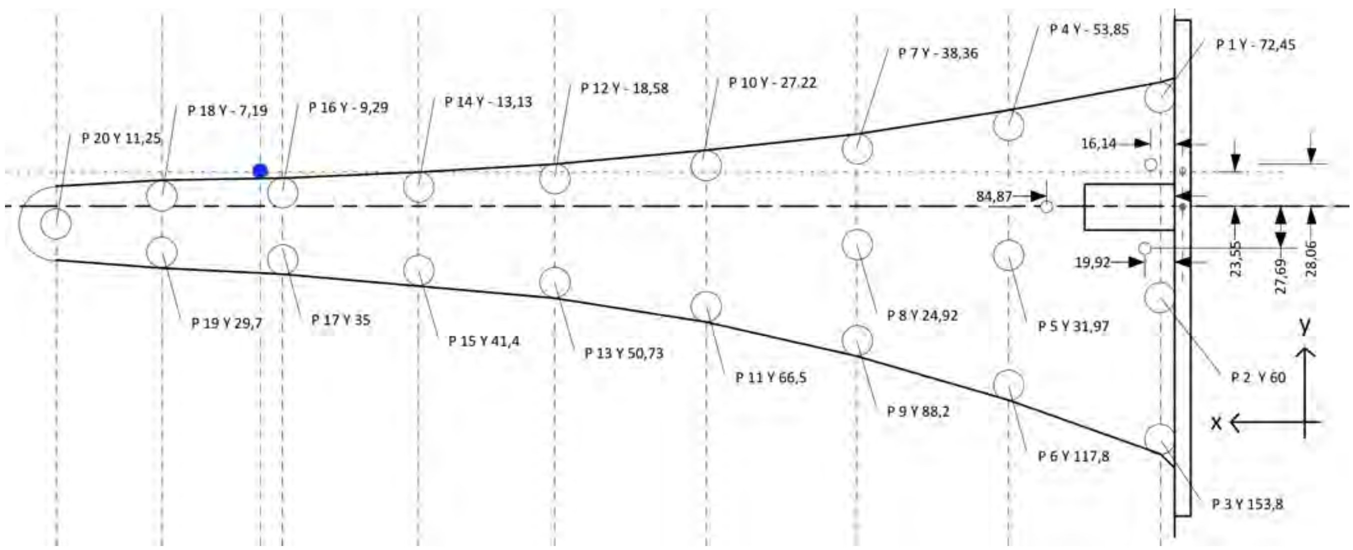
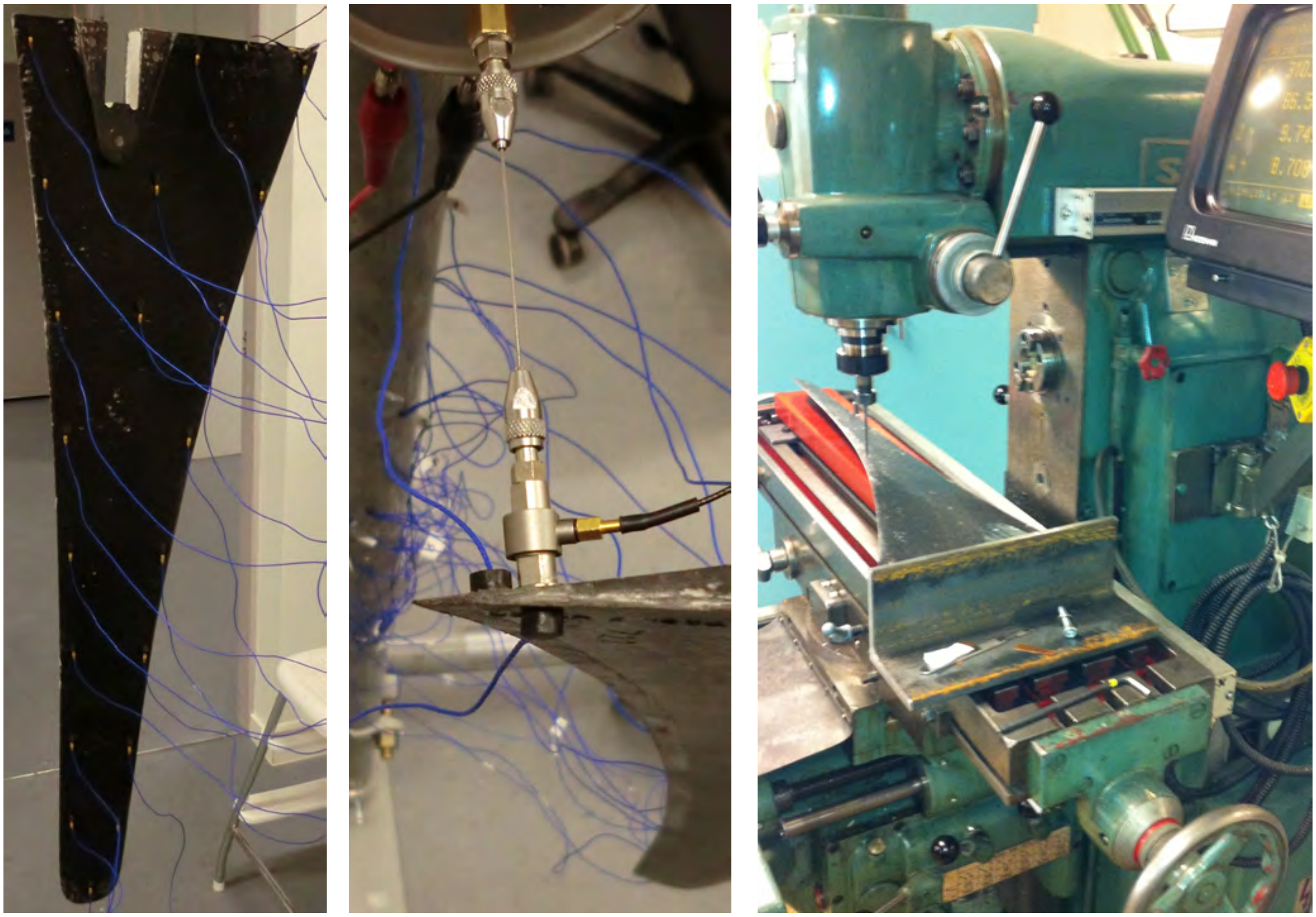


Fig. 1 Drawing of one blade with fixture (rightmost part) showing the predefined locations and their y coordinates. This was used to measure the relative height difference for seven points. Note that the z direction is perpendicular to the plane. The measurement locations were defined relative to the fixture by the point shown in red. The point in blue marks where the fixture's reference tip rested on the blade leading edge

2.2 Geometry

To characterise the geometry, a fixture was made to ensure that the same points were measured for each blade. Each blade was clamped to the fixture by two bolts. The third hole (towards the tip) could not be clamped because of variability in the hole geometry. The tip of the blade rested on a solid position so that the tip did not move during the measurements and was consistent for all blades, see blue marking in Fig. 1. One blade was used to define the x and y coordinates for all 20 locations, shown in Fig. 1. These locations were assumed consistent for all blades, although slight variation in the location markings was observed during the measurements. Furthermore, a reference point was chosen on the fixture so that for each blade the same x and y coordinates were used to measure the relative z coordinate, see red marking in 1. Seven points (P1, P2, P3, P10, P11, P18 and P19) were measured by the machine shown in Fig. 2c (which had an accuracy of approximately 1 μm). In order to be consistent throughout the measurements, a thin paper was put between the machine tip and the blade. When the tip was lowered on to the blade the paper served as an indicator of when to stop. From the measured points, an angle was calculated by knowing the distance between the two points on the same y coordinate and their relative height difference. One blade (serial

number 790) was different from the other blades in that it could only be clamped by one bolt. This separated the blade from the others by giving it slightly different test characteristics, but the blade did not behave significantly different from the other blades and was included in the statistical analysis.



(a) Test setup **(b)** Load cell **(c)** Geometry measurements

Fig. 2 (a) Vibration test setup with a blade mounted. (b) Close-up of direct accelerance sensors. The load cell is situated to the right of accelerometer 3, accelerometer 21 is behind the load cell. (c) Machine used to measure the geometry

3 Measurement Results

The geometry and mass measurements were analysed to find out if there were significant deviations between the blades and whether the difference between the four sets was significant. Further, an investigation of the dynamical properties was conducted for each blade in round 1, focusing on eigenfrequencies and the Modal Assurance Criterion (MAC), see Ewins [5]. A MAC matrix is presented for all blades, shown in Fig. 3. Also, the impact of lab testers was investigated by comparing the eigenfrequencies of multiple tests on the same blade. Note that all blades were not tested multiple times; the ones that were are specified in Table 3. Blades were ordered in sets of three as each Ampair 600W wind turbine blade set consisted of three blades. Each set was classified by a roman numeral (I, II, III, IV). The blades were classified by their serial numbers. The following serial numbers constituted the four sets: I={828, 852, 790}, II={841, 722, 819}, III={878, 877, 881}, IV={863, 962, 948}.

Note also that the six first modes (rigid body modes) are omitted and thus referring to the first mode implies the first *elastic* mode.

3.1 Geometry and Mass

In Table 1 the three cross-sectional angles for each blade are presented. Angle 1 in the table represent the angle between points P1 and P3, towards the blade base. Angles 2 and 3 represents angles between points P10, P11 and P18, P19, respectively. The

measurements imply that the blades are curved more towards the tip which can also be seen by looking at the blades. More interesting, though, is that the angles vary more towards the tip of the blades, represented by the standard deviation (σ). This can be expected from the manufacturing process where residual stresses will affect the tip to bend out of shape. The largest difference between two single blades were for blade 852 and 962; they differ around 10 degrees towards the tip.

The mass for each blade, shown in Table 1, is seen to be uniform in the groups while it differs more between the groups. This indicates that the blades were grouped in sets of three by weighing the blades. While the mass is easily seen to be grouped, the angles are harder to distinguish. In order to quantify this study, an ANOVA analysis was performed to find out whether the angles were significantly different between groups. The analysis concluded that there was no significant correlation between the angles in the groups. It other words, the blades were not grouped by their geometrical properties.

Table 1 Cross-sectional angles (degrees) and mass (gramme) for all blades, grouped in their sets. Also, mean (μ), standard deviation (σ) and the coefficient of variation (COV, %) are given for the angles and mass of all blades

Set	I			II		
Blade	828	852	790	841	722	819
Angle 1	10.09	10.32	10.15	10.06	9.94	10.19
Angle 2	21.83	21.31	21.06	20.56	22.70	23.21
Angle 3	27.79	24.97	27.15	23.62	29.78	32.83
Mass	818.80	816.60	818.20	829.60	829.00	829.50

Set	III			IV			Total		
Blade	878	877	881	963	962	948	μ	σ	*COV*
Angle 1	9.99	9.97	9.83	10.02	10.00	9.98	10.04	0.13	1.27
Angle 2	21.28	21.35	21.76	22.20	24.78	22.37	22.03	1.14	5.17
Angle 3	25.51	25.59	27.28	28.43	35.53	27.34	27.98	3.38	12.07
Mass	812.10	812.40	812.70	798.40	798.50	798.40	814.52	11.63	1.43

3.2 Modal Data

The dynamical properties studied are eigenfrequencies, presented in Table 2, and correlations between modeshapes, quantified by the MAC. The MAC matrix for all blades in round 1 is presented in Fig. 3. In Table 2 it is seen that the variation between eigenfrequencies increase with higher modes, which can be expected. The COV (coefficient of variation) is almost constant, at between 2 and 3 %. The highest standard deviation (σ) is held for the seventh mode. There are no outliers in the eigenfrequencies for data in round 1.

The MAC matrix between all 12 blades in Table 2 is shown in Fig. 3. In the figure, the four sets are separated by black bold lines. Correlations equal to or exceeding 0.95 are marked with a red border while correlations between 0.9 and 0.95 are marked with a cyan border. The green border represents values between 0.85 and 0.9. There are no anomalies and every off-diagonal value is under 0.1. Note that some correlations are low for the last mode. In particular mode 6 and 7 for blade 852 correlates poorly. The correlation is generally lower towards the higher modes.

Some frequency response functions obtained from the vibration testing, where a stepped sine input signal was used, are shown in Fig. 4. Towards the higher frequencies, greater differences were observed, as expected from previous results. Also, it can be seen that almost all modes are clearly separated from each other.

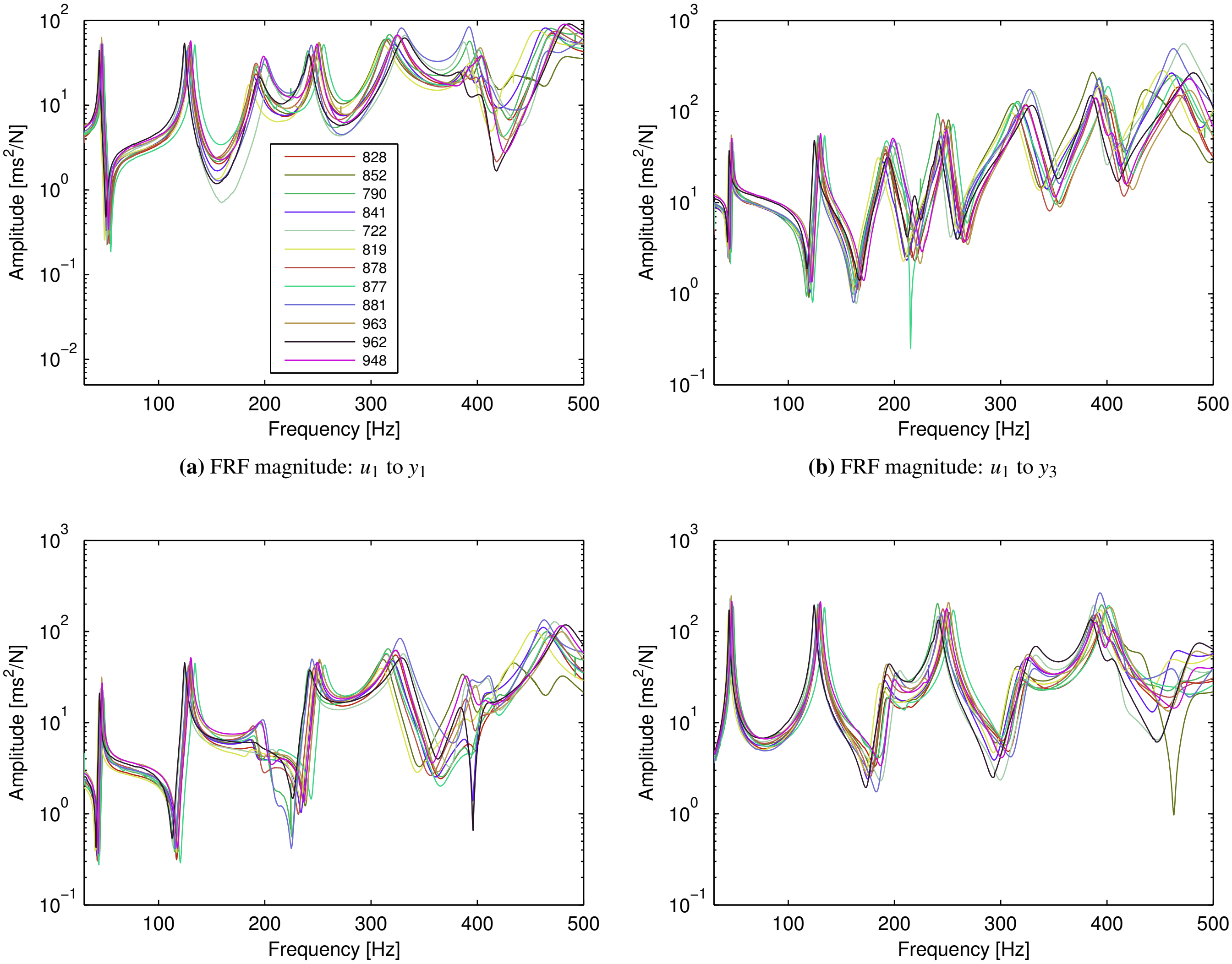
Based on results from the mass measurements regarding differences between blades in and between the sets, an ANOVA analysis was conducted to find out wether the hypothesis holds for the dynamical properties as well. The ANOVA analysis concluded that there was no significant correlation between the blades in the sets. This means that the blades were not grouped in sets of three based based on their dynamical properties.

Table 2 Eigenfrequencies (Hz) for all the blades from test round 1 grouped in their sets. Also, mean (μ, Hz), standard deviation (σ, Hz) and the coefficient of variation (COV, %) are given for each mode of all blades

Set	I			II		
Blade	828	852	790	841	722	819
Mode 1	45.60	46.48	48.43	45.16	45.18	43.58
Mode 2	129.32	128.86	128.10	128.78	124.61	128.78
Mode 3	191.18	190.76	198.77	189.26	203.54	185.17
Mode 4	248.80	251.42	240.84	248.27	241.02	247.72
Mode 5	323.99	313.92	318.07	321.68	331.27	311.74
Mode 6	399.28	388.73	394.61	392.72	388.09	393.95
Mode 7	467.65	434.33	464.76	461.95	471.34	454.03

Set	III			IV			Total		
Blade	878	877	881	963	962	948	μ	σ	*COV*
Mode 1	46.66	47.61	45.48	46.39	44.31	46.82	45.97	1.37	2.97
Mode 2	128.23	134.11	126.95	129.63	124.46	130.27	128.51	2.54	1.98
Mode 3	191.34	192.34	199.99	191.90	193.57	198.47	193.86	5.25	2.71
Mode 4	245.67	255.43	244.14	251.01	241.21	249.22	247.06	4.61	1.87
Mode 5	316.07	315.15	328.25	324.69	331.45	326.72	321.92	6.85	2.13
Mode 6	391.05	401.59	393.44	401.39	385.84	393.43	393.67	5.04	1.28
Mode 7	469.43	467.00	464.02	478.61	482.29	478.40	466.15	12.76	2.74

Fig. 3 MAC comparison for 7 modes based on tests of 12 individual blades. Blades are identified by their serial numbers. A cell with red borders marks a correlation over 0.95 while a cyan border marks a correlation over 0.9 but under 0.95. A green border marks a correlation between 0.85 and 0.9. The presented data is from tests performed in round 1



(a) FRF magnitude: u_1 to y_1

(b) FRF magnitude: u_1 to y_3

(c) FRF magnitude: u_1 to y_7

(d) FRF magnitude: u_1 to y_{20}

Fig. 4 Four frequency response functions from tests in round 1. The excitation is denoted (u_1) and the accelerometer outputs are denoted (y_1), (y_3), (y_7) and (y_{20}) where the number specifies the location on the blade, see Fig. 1. Note that the excitation is located just beside sensor 3 on the blade, see Fig. 2b for its physical location. Sensor 3 gives the direct accelerance and its frequency response function is shown in subfigure (b)

3.3 Multiple Tests

Some blades were tested two and and three times by different lab testers in order to investigate the impact lab testers had on the measurements. The results from the tests are presented in Table 3 as the mean (μ) and standard deviation (σ) between multiple tests. Also presented in Table 3 are eigenfrequency statistics for all performed tests, i.e. for blades tested once and multiple times. Although the standard deviation for the blades tested multiple times is smaller than that of the population at large, it is not insignificant. For blade 790, 828 and 772 the variation in mode 7 is high, compared to the other blades in Table 3. The highest standard deviation is found for blade 828, for each mode. Note also that blade 948 has very small and uniform standard deviations for all modes and with a uniform distribution, compared to the other blades.

One should keep in mind that the tests were performed by inexperienced lab testers, mainly by students conducting their first vibration test. Also, depending on how the stinger, seen in Fig. 2b, was configured, there was a noticeable difference in test results.

Table 3 Eigenfrequency statistics from repeated testes, given on the form $\mu \pm \sigma$ in (Hz). Also, mean (μ, Hz), standard deviation (σ, Hz) and the coefficient of variation (COV, %) are given for all tests performed on every blade

	Tested thrice				
Blade	790	841	963	962	948
Mode 1	48.35 ± 0.08	45.25 ± 0.09	46.36 ± 0.02	44.26 ± 0.05	46.83 ± 0.01
Mode 2	128.02 ± 0.18	128.97 ± 0.23	129.59 ± 0.04	124.34 ± 0.13	130.26 ± 0.01
Mode 3	198.44 ± 0.48	190.10 ± 0.92	191.75 ± 0.14	192.98 ± 0.64	198.42 ± 0.05
Mode 4	240.89 ± 0.30	248.64 ± 0.39	250.78 ± 0.21	240.96 ± 0.26	249.26 ± 0.09
Mode 5	317.12 ± 1.18	322.25 ± 0.85	324.37 ± 0.31	330.77 ± 0.72	326.69 ± 0.03
Mode 6	394.24 ± 1.08	392.92 ± 0.61	401.26 ± 0.22	385.32 ± 0.60	393.47 ± 0.12
Mode 7	463.24 ± 3.12	462.01 ± 1.76	478.18 ± 0.40	481.40 ± 1.03	478.32 ± 0.21

	Tested twice				Total		
Blade	828	722	877	881	μ	σ	*COV*
Mode 1	45.20 ± 0.55	45.09 ± 0.12	47.54 ± 0.10	45.39 ± 0.13	46.01	1.34	2.92
Mode 2	126.55 ± 3.91	124.37 ± 0.34	133.92 ± 0.27	126.80 ± 0.21	128.18	2.73	2.13
Mode 3	196.40 ± 7.39	202.91 ± 0.89	192.52 ± 0.26	199.71 ± 0.40	194.83	4.73	2.43
Mode 4	244.16 ± 6.56	240.45 ± 0.80	255.12 ± 0.45	243.68 ± 0.66	246.28	4.89	1.98
Mode 5	325.16 ± 1.67	330.70 ± 0.80	314.98 ± 0.24	328.18 ± 0.09	323.21	6.03	1.87
Mode 6	392.82 ± 9.13	387.60 ± 0.70	401.22 ± 0.53	392.90 ± 0.75	393.25	5.26	1.34
Mode 7	471.94 ± 6.06	473.75 ± 3.41	467.31 ± 0.44	464.48 ± 0.65	469.32	10.64	2.27

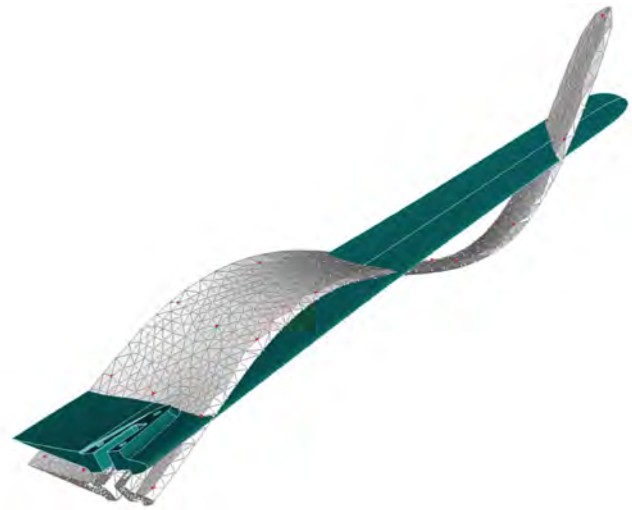
4 FEM Correlation

The course, in which the work for this paper was conducted, included correlating and calibrating a FE model by MAC values and eigenfrequencies, which will be presented here. Apart from this, the course content consisted roughly of; pretest planing, where emphasis was put on the method of effective independence, see [7]. Vibration testing, where students used the available test equipment, and system identification, where students used the System Identification Toolbox in MATLAB.

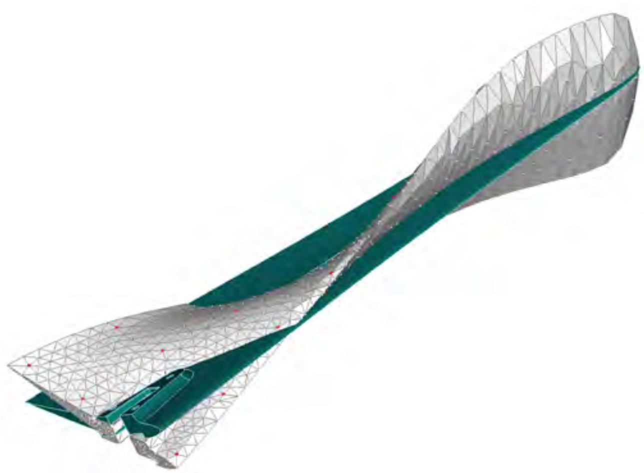
The outer geometry of the blade was provided by the SEM Substructuring focus group, from which a FE model was built in FEMAP and analysed using NX NASTRAN. The FE model had to be simplified as the material of the blade was unknown and some standard initial material had to be guessed. The material parameters were chosen from from [8]. Further, simple isotropic shell elements were used to model the skin of the blade and no solid core was used. In order to find improved material parameters, latin hypercube sampling was used to find 10 sample locations. Latin hypercube sampling is further explained in [9]. These 10 samples were then mapped to a beta distribution which was assumed to describe the material properties. Thus, 10 new independent values of Young's modulus and shear modulus were obtained. The 10 new models were then correlated with the measurements and compared to the initial model. This method was used as there was no time to perform a more advanced optimisation. Also, the FE model was simplified so much that more advanced optimisation methods would probably not have yielded significantly better results.

In Fig. 5 the best fitted model is correlated with blade 772. It can be seen that quite good correlation is achieved for the first four modeshapes. The two first modes have MAC values over 0.95. The eigenfrequencies also correlated reasonably well for the first four modes as seen in the subfigures of Fig. 5.

(a) First elastic modeshape, 48.32 Hz (measured mean: 46.01 Hz)



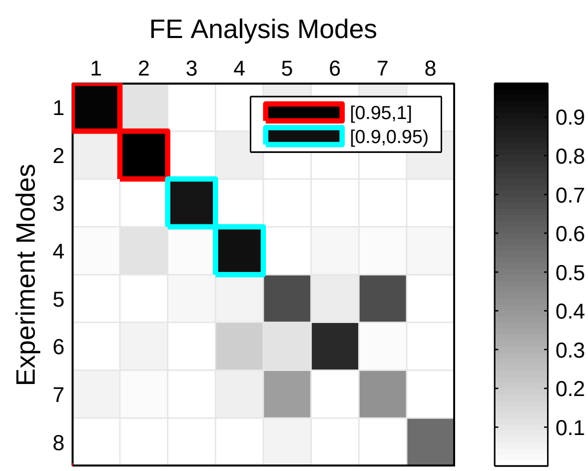
(b) Second elastic modeshape, 128.59 Hz (measured mean: 128.18 Hz)



(c) Third elastic modeshape, 183.75 Hz (measured mean: 194.83 Hz)

(d) Fourth elastic modeshape, 230.75 Hz (measured mean: 246.28 Hz)

(e) Fifth elastic modeshape, 312.30 Hz (measured mean: 323.21 Hz)



(f) MAC between blade 772 and FE model

Fig. 5 The first five elastic modeshapes for the FE model are shown from (a) to (e). The measured mean is the same as in Table 3. The red dots are sensor locations as specified in Fig. 1. (f) The MAC matrix

5 Conclusions

The Ampair 600W wind turbine system has been chosen as a benchmark test bed for the SEM Substructuring focus group, with machines already set up in several countries. To ensure a meaningful comparison of these, a quantification of the expected variability of the turbine system's integral components is of great interest. This study, focuses on the variability of the blades.

A series of measurements on 12 blades from the wind turbine have been performed as part of the course, Structural Dynamics Model Validation, given at Chalmers University of Technology. ANOVA results have been used to confirm that the blades come in weight-matched sets, but also that weight-matching does not imply a similar matching of geometrical or dynamical properties.

The measured spread between the different blades was considerable. Measured pitch angles at different positions along the length of the blade showed increasing Coefficients of Variation (COV), from 1.27 % at the base to 12 % at the tip. The COV for the seven eigenfrequencies measured for each blade was stable, at around 3 % for all modes.

Different lab testers were responsible for the vibration testing of different blades. To investigate the effect of this on the modal data, a few blades were tested multiple times, by different people. The spread resulting from a change of lab testers varied between the different blades, ranging from almost insignificant to standard deviations comparable to that between blades.

A spurious mode was identified in the measurements. The mode was not consistent with either the MAC correlation nor the eigenfrequencies, not even for the same blade tested multiple times in seemingly well-performed tests. This mode was removed as it was caused by either imperfect drive rod assembly, see [5], or by the system identification process.

Finally, a linear and isotropic FE model with shell elements was built. It was possible to calibrate the model so that the first four modes had a MAC-correlation exceeding 90 %. To further improve the FE model, an orthotropic material model must be used and the core material must be modelled. Such a model is presented in [10].

References

[1] Friswell, M. I., Mottershead J. E., Finite Element Model Updating in Structural Dynamics, Kluwer Academic Publishers, Dordrecht, 1995.

[2] Mayes, R., ”An Introduction to the SEM Substructures Focus Group Test Bed The Ampair 600 Wind Turbine”, Proceedings of the 30th IMAC, Jacksonville, (2012), 61-70.

[3] Harvie, J., Avitabile, P., ”Comparison of Some Wind Turbine Blade Tests in Various Configurations”, Proceedings of the 30th IMAC, Jacksonville, (2012), 73-79.

[4] Nurbhai, M., Macknelly, D., ”Modal Assessment of Wind Turbine Blade in Preparation of Experimental Substructuring”, Proceedings of the 30th IMAC, Jacksonville, (2012), 71-72.

[5] Ewins, D. J., Modal testing: theory and practice, Research Studies Press, Baldock, 2000.

[6] McKelvey, T., Akcay, H., Ljung, L., Subspace-based multivariable system identification from frequency response data, IEEE Transactions on Automatic Control 41 (7), (1996), 960-979.

[7] Kammer, D. C., Sensor set expansion for modal vibration testing, Mechanical Systems and Signal Processing (14), 2005, 700-713.

[8] Sundström, B., Handbok och formelsamling i hållfasthetslära [Handbook and formulas in solid mechanics], Department of Solid Mechanics, KTH, Stockholm, 1988, *in Swedish*.

[9] McKay, M. D., Beckman, R. J., Conover, W. J., A Comparison of Three Methods for Selecting Values of Input Variables in the Analysis of Output from a Computer Code, Technometrics (21), (1979), 239-245.

[10] Johansson, A. T., Lindholm, C-J., Khorsand, M., Abrahamsson, T., ”Modeling and calibration of small-scale wind turbine blade”, Proceedings of the 31th IMAC, Garden Grove, (2013). In Press.